DTX 104

Hamilton Standard DIVISION OF UNITED AIRCRAFT CORPORATION UA

INTERNAL CORRESPONDENCE

CD 691 KHV

Please address answer to Mail Stop No 1A-3-2

May 30, 1975

Memorandum to: Mr. R. Emmons

cc: Messrs. J. Burger, R. Goodman, K. Harner, J. Patrick

From: Mr. M. Spadafora

Subject: L-1011 APU Surge Control Redesign Feasibility Study

References:
1) CD 428, "L-1011 APU Surge Control Dynamic Analysis" 8/18/70
2) JBG-PA861-L-1238, "L-1011 APU Pneumatic and Electric Transients 7/5/73

Introduction

During 1973, the performance of the APU surge control hardware, as determined by HSD testing and field service performance, was not doing an adequate job of preventing engine compressor surge when subjected to various types of ECS flow demand reductions. The primary cause, it was understood, was shifting of the rate sensor threshold due to hardware problems. To minimize the shifting problems, Project Engineering initiated a Mechanical Design Group Effort to improve the hardware. In addition, they initiated a Control Dynamics Group effort to define a new controller concept which would provide significant improvements in both steady-state accuracy and transient performance.

This report has been written to provide a summary of the Control Dynamics Group redesign effort performed in 1973 on the L-1011 APU surge control. The summary includes the performance characteristics of this new controller, and compares them to the 1973 configuration system. Also included in the nonlinear computer simulation results are the transient results for the flow disturbances defined by Lockheed in Reference 2. The latter disturbances have not been evaluated in any previous surge control study.

The concept that was decided upon utilizes a pneumatic controller with actuator rate feedback. The rate feedback allows the steady-state accuracy on compressor corrected flow to be improved significantly while maintaining comparable dynamic stability. Since transient error in compressor corrected flow (a measure of compressor surge potential) is the primary concern, three additional steps were taken to minimize this error for the typical system disturbances:

SUND 000268

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TABLE II

(New Design Concept)

A_1	Rate feedback sensing diaphragm area	= 2. (in^2)
A_2	Pressure signal sensing diaphragm area	= 2. (in^2)
A_H	Actuator high pressure side piston area	– 4.41 (in^2)
A_{IN}	Servo supply pressure orifice area	– .0007 (in^2)
A_{SE}	Actuator servo pressure side piston area	– 7.07 (in^2)
A_T	Downstream duct restriction area	– 13.27 (in^2)
C	Discharge coefficient for downstream restriction, flapper, supply orifice, and anti-saturation poppet	– .7
D	Surge valve diameter	– 4.84 (in)
$DAMP_A$	Actuator damping coefficient	– 2. $(\frac{lb}{in/sec})$
D_F	Flapper diameter	– .15 (in)
D_P	Anti-saturation poppet diameter	– .05 (in)
D_{R1} (1)	Rate feedback laminar restrictor dia.	– .021 (in)
D_{R2} (2)	Signal compensator laminar restrictor dia.	– .05 (in)
FLOAD	Surge valve aerodynamic load referred to the actuator (Reference 1)	
FRICTA	Nominal value of actuator friction from test data	– 14.1 (lb)
K_S	Flapper lever spring rate	– 36. (lb/in)
L_1 (3)	Lever length to feedback pressure sensing diaphragms	
L_2 (3)	Lever length to signal sense diaphragms	
L_F (3)	Lever length to flapper and preload spring	
L_{R1} (1)	Rate feedback restrictor length	– 2. (in)
L_{R2} (2)	Signal compensator restrictor length	– 3. (in)
MA	Actuator mass	– 1.3×10^{-3} $(\frac{lb}{in/sec^2})$
P_3	Engine bleed pressure	
P_{AMB}	Ambient pressure	– 14.7

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TABLE II

(Continued)

P_D	Surge valve downstream pressure	
P_{FB} (4)	Rate feedback signal pressure	
P_{G2}	Sensed static pressure signal	
P_{G3}	Sensed total pressure	
P_{LAG1}	Lagged rate feedback signal pressure	
P_{LAG2}	Lagged static pressure signal	
P_{SE}	Actuator modulated pressure	
PU	Load compressor discharge pressure	
TU	Load compressor discharge temperature	353°F (max. mode)
V_D	Duct pressure volume downstream of the surge valve	
V_{SE}	Actuator servo volume	
V_{R1} (1)	P_{lag1} chamber volume	2. (in^3)
V_{R2} (2)	P_{lag2} chamber volume	2. (in^3)
V_U	Compressor downstream pressure	1.75 to 50. (ft^3)
W_{ECS}	Outflow from the compressor downstream pressure volume to ECS packs or ATM's	
W_{SV}	Flow through the surge valve	

Notes:

(1) This sizing for D_{R1}, L_{R1}, and V_{R1} was used to obtain a lag time constant of .1 seconds at sea level.

(2) This sizing for D_{R2}, L_{R2}, and V_{R2} was used to obtain a lag time constant of .01 seconds at sea level with the IGV full open (max. mode)

(3) To get the desired flow loop gain, $\frac{L2}{LF} = \frac{L1}{LF} \times 3.6$

(4) The rate feedback pressure gain characteristic used to optimize the flow loop gain characteristic is shown in Figure 8.

SUND 000275

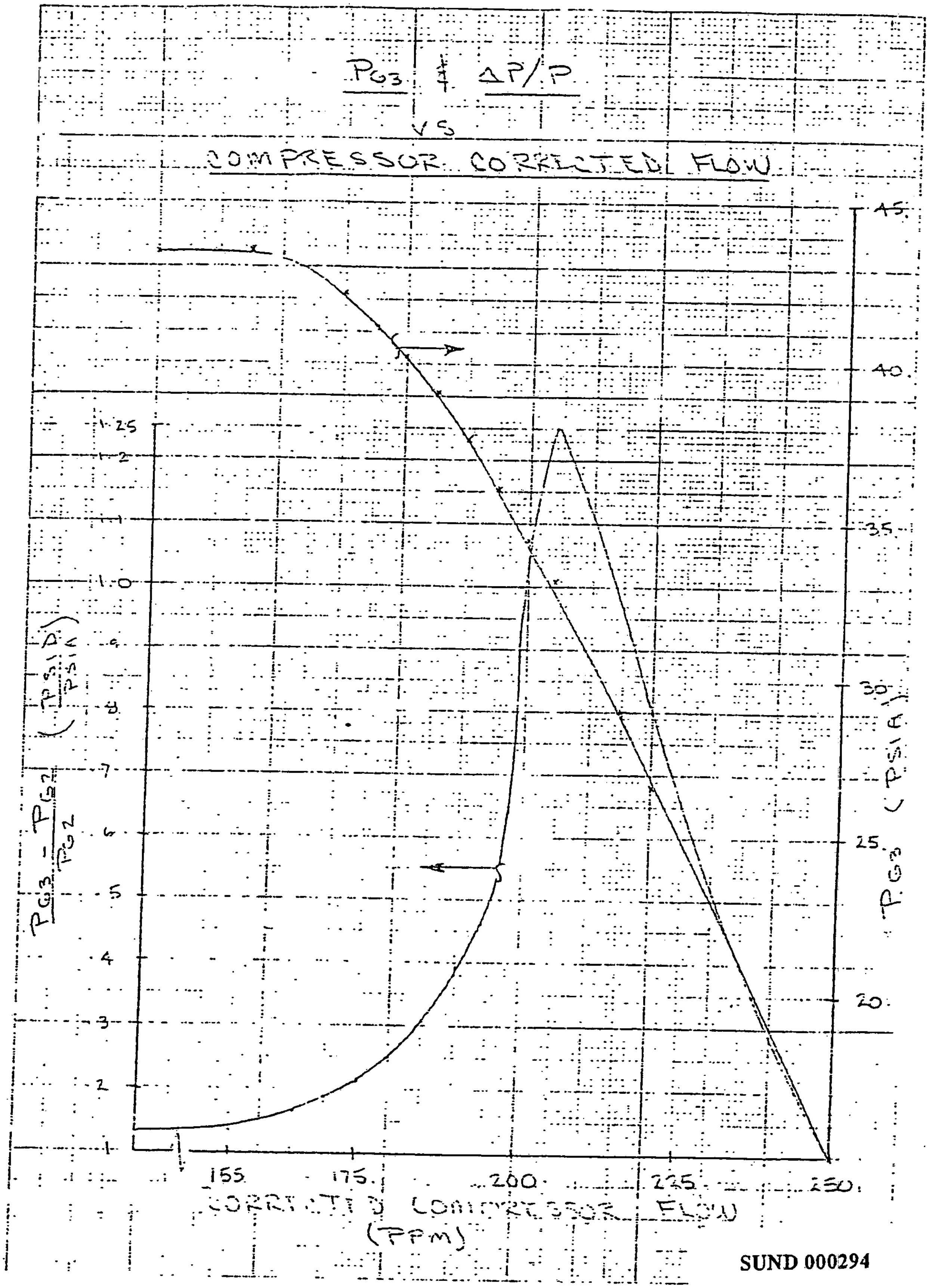

FIGURE 9

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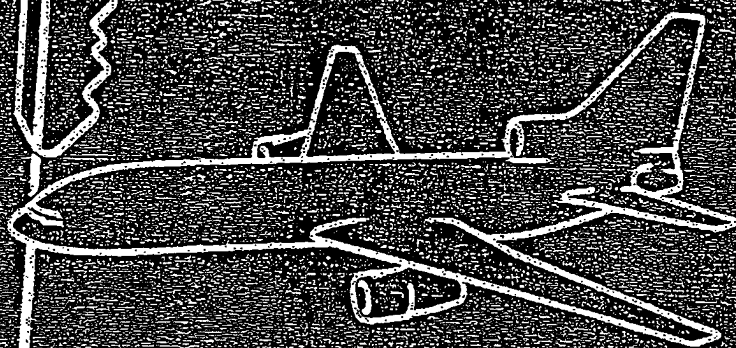
DTX 105

H. HIPSKY

AUGUST 15, 1975
VOLUME 2

MASTER KEY

L1011 TRISTAR
AUXILIARY POWER UNIT



- SYSTEM DESCRIPTION/OPERATION
- TROUBLE SHOOTING

ST6L-73 ENGINE

LOAD COMPRESSOR SYSTEM

CONTROLS AND INDICATOR INTERFACES

APU CONTROL SYSTEM

APU COMPARTMENT INTERFACES

APU TROUBLE SHOOTING

HAMILTON STANDARD Division of UNITED TECHNOLOGIES

SUND 000431

Honeywell KIMMIE EXHIBIT 5
FOR I.D. 12/9/05

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Load Hardware Gearbox Description

The load hardware gearbox (Figure 2-5) consists of a gearbox, load compressor, and cooling air fan. The gearbox secured to the engine housing, serves as a power distribution source to drive the cooling air fan, and electric ac generator.

The load compressor is a centrifugal air compressor mounted directly on the aft end of the main gearbox drive shaft. The compressor inlet supports the inlet guide vane assembly. A bypass port at the compressor outlet provides mounting for the surge valve.

The load compressor housing diffuser forms a venturi chamber which has four pressure pickoffs; P_{S0}, P_{S1}, P_{S2} and P_T. The venturi chamber is a divergent duct to subsonic flows and there is a progressive pressure rise from P_{S0} to P_T. If the flow becomes supersonic the venturi chamber acts as a convergent duct and there is a progressive pressure loss from P_{S0} to P_T.

Later models of the load compressor have an adjustable orifice attached to the compressor housing over the P_T port which adjusts the characteristics of the load compressor to be compatible with any surge valve control. This orifice adjustment was initially set during manufacture and finally set during APU testing. This is not normally a field adjustment.

The cooling air fan is a seven-bladed, mixed-flow type axial fan encased in a molded plastic housing which attaches directly to the gearbox. The gearbox also provides a mounting pad and adapter for attachment of the ac generator.

Load Hardware Gearbox Operation

Rotational power is transmitted from the free turbine shaft through a quill shaft to the load compressor. Bevel gears, lubricated by a spray bar, split the torque path so that the free turbine drives to the load compressor and ac generator simultaneously. The cooling air fan is spur driven from the generator drive gear and the cooling air fan shaft incorporates a reverse rotation-locking clutch.

The load compressor draws air through the inlet guide vane assembly, compresses the air and supplies this air to aircraft pneumatic system supply ducts and to the APU surge valve.

The cooling air fan (Figure 2-6) provides air for APU compartment ventilation, engine exhaust shroud cooling, and oil-to-air heat exchanger cooling to cool generator and engine oils.

Load Hardware Gearbox Leading Particulars

- Gearbox Input 33,000 rpm, constant

SUND 000499

DTX 182

AIRESEARCH MANUFACTURING COMPANY

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA

OFFICE MEMO

IN REPLY REFER TO:
CA:JCC:0021:031578

DATE: March 15, 1978

TO: F. E. Johnson DEPT. 93-271-503-3V

FROM: J. C. Clark DEPT. 93-082-503-3G EXT. 4854

COPIES TO: R.P. Craig, D. Romero, R.F. Stokes, J.R. Sullivan, CHRONO

SUBJECT: F-18 SURGE CONTROL SYSTEM DEVELOPMENT SUMMARY

REFERENCES:

1) GTC36-200 Surge Control Loop Dynamics CA:JCC:0001:102076
2) F-18 Surge Valve Size Verification CA:SDA:0005:112476
3) Flow Sensor Signal CA:DPT:0271:051677
4) F-18 Surge Valve CA:JCC:0010:070677
5) F-18 Surge Valve Modifications CA:JCC:0013:072777
6) GTC36-200 Pneumatic Test Plan CA:JCC:0015:081577
7) Butterfly Aerodynamic Torque 4162:RJD:0517:1201-7
8) Surge Valve Calibration on F-18 SP2 Engine
9) Weekly Status Report, February 19, CA:RFS:0253:022078
10) Surge Bleed Control Logic CA:RFS:0194:040177

I. SUMMARY

Development testing on the F-18 surge valve P/N 109768 is nearly complete. The only open item is calibration (see references 8 and 9). A limited amount of test data has shown variation in flow sensor signals on different engines. With the increased surge margin seen on engine #4 (12% S.M., which is the specified value) and the valve capability to operate with (3% S.M.) there will be "room" for flow sensor variation. On January 28, a production version of the surge valve passed the stability and transient tests while running the engine ATP. Therefore, the valve configuration has been finalized.

A sample of a typical surge valve response is shown below, Figure 1. The trace shows a response to a closing of a simulated aircraft valve. There is no undershoot in delta P and only about ½ PSI overshoot in P_T. Also, the initial opening time of the valve is very fast and the steady state is very stable.

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AS 085893

II. SYSTEM OPERATION (DEVELOPMENT VALVE)

Logic Derivation

A description of the surge valve logic is given in reference 10. The crux of surge control logic is to find one parameter that will indicate incipient surge irrespective of speed and inlet guide vane angle (IGV). The derivation of the logic starts with approximating the surge line with a linear equation relating compressor pressure ratio and corrected inlet flow.

$$\frac{P_{02}}{P_{01}} = K \frac{W\sqrt{T_1}}{P_{01}}$$

Subscript 1 is for inlet;
2 is for discharge.

This is a good approximation and holds for varying speeds and IGV angles. For different values of K there is a family of lines at the surge line K is maximum. Therefore, the desired parameter is K.

This equation is then written in terms of corrected discharge flow:

$$\frac{P_{02}}{P_{01}} = K \frac{W\sqrt{T_1}}{P_{01}} = K \frac{W\sqrt{T_2}}{P_{02}} \frac{P_{02}}{P_{01}} / \sqrt{\frac{T_2}{T_1}}$$

Observe that: $\frac{T_2}{T_1} = 1+ \frac{T_2 - T_1}{T_1} = 1+ \frac{\text{Delta T}}{T}$

Cancelling the pressure ratio terms gives:

$$1 = K \frac{W \quad T_2}{P_{02}} / \sqrt{1+ \frac{\text{Delta T}}{T}}$$

Solving for K gives:

$$K = \sqrt{1+ \frac{\text{Delta T}}{T}} / \frac{W\sqrt{T_2}}{P_{02}}$$

The value of $\sqrt{1+ \frac{\text{Delta T}}{T}}$ is nearly constant; therefore, the quantity that must be measured is $\frac{W\sqrt{T_2}}{P_{02}}$. This can be done with a flow sensor consisting of a total and static probe.

DTX 202

FORM GT734-6F

GARRETT TURBINE ENGINE COMPANY
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA
OFFICE MEMO

Refer to:
CA:JCC:0090:121683

DATE: December 16, 1983

TO: John Dannan DEPT. 93-273-503-3Y

FROM: Jim Clark DEPT. 93-082-503-36 EXT. 4854

COPIES TO: P. Bliklen, K. Davis, C. Lewis, R. Stokes, J. Volkmann, Chrono

SUBJECT: TEST RESULTS OF GTCP85-1000 DIFFUSER FLOW SENSOR

REFERENCE: GPG SURGE CONTROL VALVE (SCV) PROBLEM STATEMENT, CA:CPH:0012:102683

PROPRIETARY

INTRODUCTION:

A surge control system that is being investigated for the GPG will use static pressures, located in the diffuser, to detect incipient surge. The pressures are located on the vane leading edge high pressure P_1, leading edge low pressure P_2 and the diffuser exit P_3. It was expected that surge will occur when the ratio $(P_3 - P_1)/(P_3 - P_2) = 0.3$.

Another approach that is being considered will use the diffuser total pressure and exit static pressure to form the ratio $(P_T - P_S)/P_T$. This will then be the parameter used to detect surge.

Engine demo A had a diffuser instrumented to investigate the feasibility of these approaches. Testing on this engine showed that the $\Delta P/\Delta P$ approach was the best scheme.

DISCUSSION

The test plan consisted of measuring signal levels at several performance and surge points at various speeds. This was done to see if there was "room" between the surge points and the performance points to operate a surge valve.

A diffuser was used that had static pressures instrumented in five passages, as shown in figure 1. Also, the total pressure was measured by a kiel probe located in the deswirl area. These pressures were measured by one transducer using a scanning valve. This eliminated tranducer errors when calculating delta P.

The bleed flow was measured by the orifice section in the cell. Compressor inlet flow was calculated using the total and static pressures in the diffuser exit. Knowing these pressures and the effective area of the diffuser the flow was calculated from the mach tables, see attachment A for calculation of effective area.

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Dep. Ex. 22

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RMD AS 000070

It is estimated from past experience that the valve droop would be about 1.5 psi. The setpoint tolerance will be about $\pm$ .1 psi. The slope is a function of diaphragm areas. This tolerance could be held to about $\pm$ .1 psi. Without doing a transient analysis it is hard to calculate the transient margin. However, a good guess would be about 1 psi. This gives a total band of 2.9 psi.

As seen on figure 11 there is a considerable spread in the surge points. If the sensing holes could be located within a tolerance of 0.020 inch and using the sensitivity calculated from figure 9 the spread of the surge points could be made to fall within the shaded region of figure 12.

Combining the surge valve tolerances with flow sensor tolerances it seems feasible that the $\Delta P/\Delta P$ control would work.

The other approach of using $\Delta P/P_T$ was also investigated. Figure 13 shows a plot of diffuser exit pressure delta P vs P_T. Because of the ΔP is smaller this approach is not as good as the $\Delta P/\Delta P$ approach. The delta P can be increased by using the leading edge suction pressure to form the ΔP. However, this forms a nonlinear surge line which makes it hard to design a valve.

CONCLUSION

The information gained during this test showed that the $\Delta P/\Delta P$ type surge control system is a promising concept. The signal levels were found to be large and the ratio of $\Delta P/\Delta P$ is a good indication of incipient surge.

Because the signals are large even at low speeds it maybe possible to use such a system to trim fuel during starts. This would be better than an open loop surge schedule. This is an important idea to keep in mind for future programs. It would also be an interesting approach to use on a gas generator where rapid accelerations are required. This concept may also lend itself to load compressores that have inlet guide vanes. The use of the $\Delta P/\Delta P$ would do away with scheduling the surge valve setpoint with guide vane angle.

So far two compressors have found to surge at $\Delta P/\Delta P = 0.3$. It is expected that all compressors with vanes of the same shape will surge at the same ratio regardless of the compressor size. This is an important point because the same basic surge valve could be used for many different engines. Even if they are load compressor types or integral bleed engines.

Jim Clark

Jim Clark
Systems Analysis & Definition
Controls & Accessories Project

JCC:lj
Enclosures: 1. Calculation of diffuser effective area
2. Summary of data

DTX 203

FORM GT334-6F

GARRETT TURBINE ENGINE COMPANY
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA
OFFICE MEMO

Refer to:
CA:JCC:0113:050185

DATE: May 1, 1985

TO: Larry Farina DEPT. 93-220/503-3AR

FROM: [redacted]/Rich Stokes DEPT. 93-082/554-D EXT.4854/2298

SUBJECT: SELECTION AND OPERATION OF THE GTC131 SURGE CONTROL SYSTEM

COPIES TO: M. Adams
J. Clapp
C. Hickey
S. La Croix
G. Lukert
B. McCarthy
R. McGinley
J. McLeod
G. Perrone
L. Smalley
D. Thomason
Chrono

PROPRIETARY

INTRODUCTION

Because of mechanical problems in locating a flow sensor in the bleed duct and system accuracy considerations, a different type of surge control logic will be used on the GTC131.

The conventional surge control system airflow measurement technique originally proposed for the GTC131 was the same as currently being used on the GTCP331 APU. This system requires a total and static pressure measurement of the entire load compressure discharge airflow. The restricted bleed duct configuration of the GTC131 does not permit this.

In addition, the tolerances of the GTCP331 system requires a 12 percent margin between the operating points and the surge line. The GTC131 will only have a 10 percent margin.

As a result, a unique new surge control system bleed air measurement technique will be used. The operation and accuracy of this system are discussed below.

OPERATION

The main assumption in the development of the GTC131 logic is that surge will be caused by aerodynamic stall in the diffuser. Airflow will stall on the diffuser vanes at a unique angle of attack. The angle of attack can be sensed by examining the pressure distribution along the vane.

Figure 1 shows the principle of how the angle is detected. The differential-pressure coefficient ($P_T - P_S/0.5\ \rho V^2$) is shown plotted along the profile of the vane for different angles of attack. On the plot, Position B is on the pressure side of the vane and Position A is on the suction side. The figure shows how pressure distributions vary with angle of attack. The pressure coefficient for the pressure side is

$$\frac{P_T - P_B}{0.5\rho V^2}$$

and the value for the suction side is

$$\frac{P_T - P_A}{0.5\rho V^2}$$

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RMD AS 000092

PROPRIETARY

leading edge at about 10 percent of the vane cord. At this position the pressure ratio $P_T - P_S / P_T = 0.16$. On the suction side, the pressure gradients at the leading edge are small. Therefore, a sensing hole at the leading edge would be a good location. At this position the pressure ratio is $P_T - P_S / P_T = .35$. If the static pressure holes are located at these positions then the ratio of $\Delta P_L / \Delta P_H$ would be given by

$$\frac{\dfrac{P_T - P_{SB}}{P_T}}{\dfrac{P_T - P_{SA}}{P_T}} = \frac{.16}{.45} \doteq 0.35 = \frac{\Delta P_L}{\Delta P_H}$$

TYPE IN 4
.35 .35

Figure 6 shows a sketch of where to locate the holes on the load compressor first stage vane. This location is recommended for the first engines because design of engine hardware will be taking place before rig test data will be available.

ACCURACY

Using the data from the GTCP85 and F109 tests, a tolerance analysis was made.

The accuracy of the new diffuser ΔP/ΔP system is shown as part of the system trade-off chart, Table I. The electronic version is projected to work with a 8.5 percent surge margin and is the only system to meet the 10 percent requirement. Since the new system eliminates the need for IGV position feedback, a purely mechanical configuration was also evaluated. Unfortunately, the mechanical system with its associated proportional control and droop required a 13 percent surge margin. This did not meet the 10 percent requirement so the improvement in weight, cost, and MTBF could not be realized.

System accuracy was calculated by determining the effect of individual components and taking the square root of the sum of the squares of all errors $(\Sigma x_i^2)^{1/2}$. For comparison, the accuracy of other surge control systems is presented in Table II.

The ΔP/ΔP system accuracy can be improved by calibrating the ΔP transducers prior to start. This can be done because the value of the transducers should be zero prior to starting. This is an advantage over the 331 type system which uses a Ptotal transducer that measures absolute pressure. This transducer can't be calibrated because its desired output is unknown. Another advantage is in failure modes. If one of the transducers should fail open or zero output, then the system could still run using the other transducer to measure flow. Although the performance would have to be down rated.

DTX 204

GARRETT TURBINE ENGINE COMPANY
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA
OFFICE MEMO

Refer to:
CA:ERG:0018:092085

DATE: September 20, 1985

TO: Rich Stokes DEPT. 93-082/554D

FROM: Ed Goff DEPT. 93-082/554D EXT. 4325

COPIES TO: M. Adams, G. Ando, P. Ardans, J. Clark, B. Minshall, K. Steen, S. Stohlgren, J. Sullivan, Chrono

SUBJECT: GTCP36-300 SURGE CONTROL SYSTEM DYNAMIC ANALYSIS AND DESIGN

REFERENCES:
1. Memo CA:ERG:0010:012385
2. Memo CA:ERG:0017:072385
3. MBB Telex FSNO. 4.504, 09/11/85
4. Surge Control Valve Procurement Specification 31-5833

INTRODUCTION

Dynamic analysis and design of the GTCP36-300 surge control loop is complete. This activity has yielded baseline design of the surge control system which should be a very good start for laboratory testing, and has provided an understanding which will be extremely valuable for finalizing the design in the laboratory. Analytical and numerical methods have been used to design the controller gains, lead-lag compensator, the orifices and volumes which filter the pressure signals, and other control logic. The system was kept as close as possible to the GTCP331 configuration, but some changes were necessary to accommodate the different dynamics of the new compressor, surge valve, flow sensor, and aircraft ducting. All of these changes are in software except for a change in the total pressure orifice leading to the volume chamber. At present, no significant problems are foreseen in the area of surge control system dynamics. (The area of steady-state tolerance versus surge margin at MES remains troublesome.)

Chronologically, there were three major steps of the design, which are documented in the following sections. First, a detailed, nonlinear mathematical model of the surge control system was devised. Second, a simplified linear model was used along with frequency response analysis to provide a stable initial controller design. Third, the initial controller design was used in a full, nonlinear digital computer simulation. With both stability and transient response criteria in mind, the design was refined with the computer simulation. The following documentation does not include iterations which refined the design; each section shows only the final design.

DETAILED MATHEMATICAL MODEL

A schematic of the surge control hardware is shown in Figure 1; the configuration is similar to the GTCP331. The structure of the full mathematical model is shown in the block diagram in Figure 2, and the parameter values and details of the model can be seen in the computer program listing in Appendix A. Some notes about the model are listed below:

- When the MES mode signal goes off, the integrator is reset to zero (open valve) and held there for two seconds.

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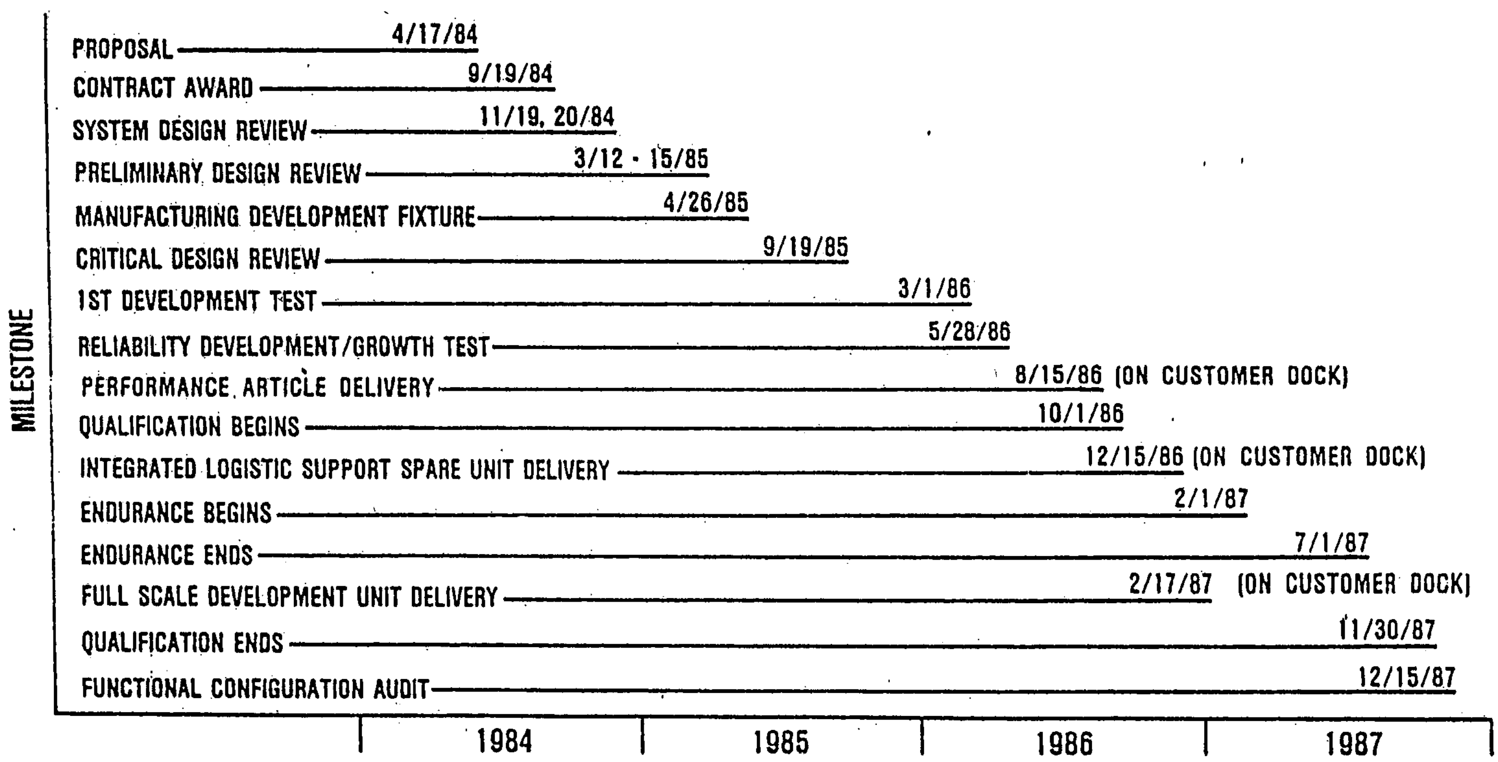
RT
JIM CLARK

GTC131 APUS PROGRAM MILESTONES



GARRETT

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7

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DTX 210

AlliedSignal

GTCP331-350 APU

GAPD/TURBOMECA COORDination MEMO

TO: H. VIGNAU

FROM: ED GOFF/ JOHN ZIMMERER

PAGE 1 OF 2

MEMO NO.: [redacted]

DATE SENT: 5-4-89

REPLY BY:

SUBJECT: SELECTION OF STATIC PRESSURE PICKUP FOR SURGE CONTROL

REFERENCE: T0124

NOTE
The following referenced information
is ☒
is not ☐ considered "PROPRIETARY"
by the originator.

☒ REQUEST ☐ INFORMATION

☒ REPLY TO: T0124

We agree that a static pressure pickup just downstream of the diffuser throat is much better than a hole in the throat. This is true not only for the reasons you mentioned, but also because with holes in the throat, the ΔP measurement can actually be negative as we have seen during engine testing; this can damage the ΔP transducer.

Please use location number 29 for the static pickup holes. Location 29 is better than any location further upstream because the corrected flow is a unique function of $\Delta P/P_t$ (within the practical range of flows) and $\Delta P/P_t$ increases with corrected flow. Location 29 is better than any location further downstream because the ratio of ΔP at the critical part of the control schedule to the highest possible ΔP is as large as possible, allowing the best accuracy of the transducer.

APPROVED BY: Charles D. Kyler

DATE: MAY 4, 1989

DISTRIBUTION TO: TURBOMECA

T. Burnett	C. Kyler	J. Randall
L. Campbell	B. Macarez	B. Stringham
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AS 009522

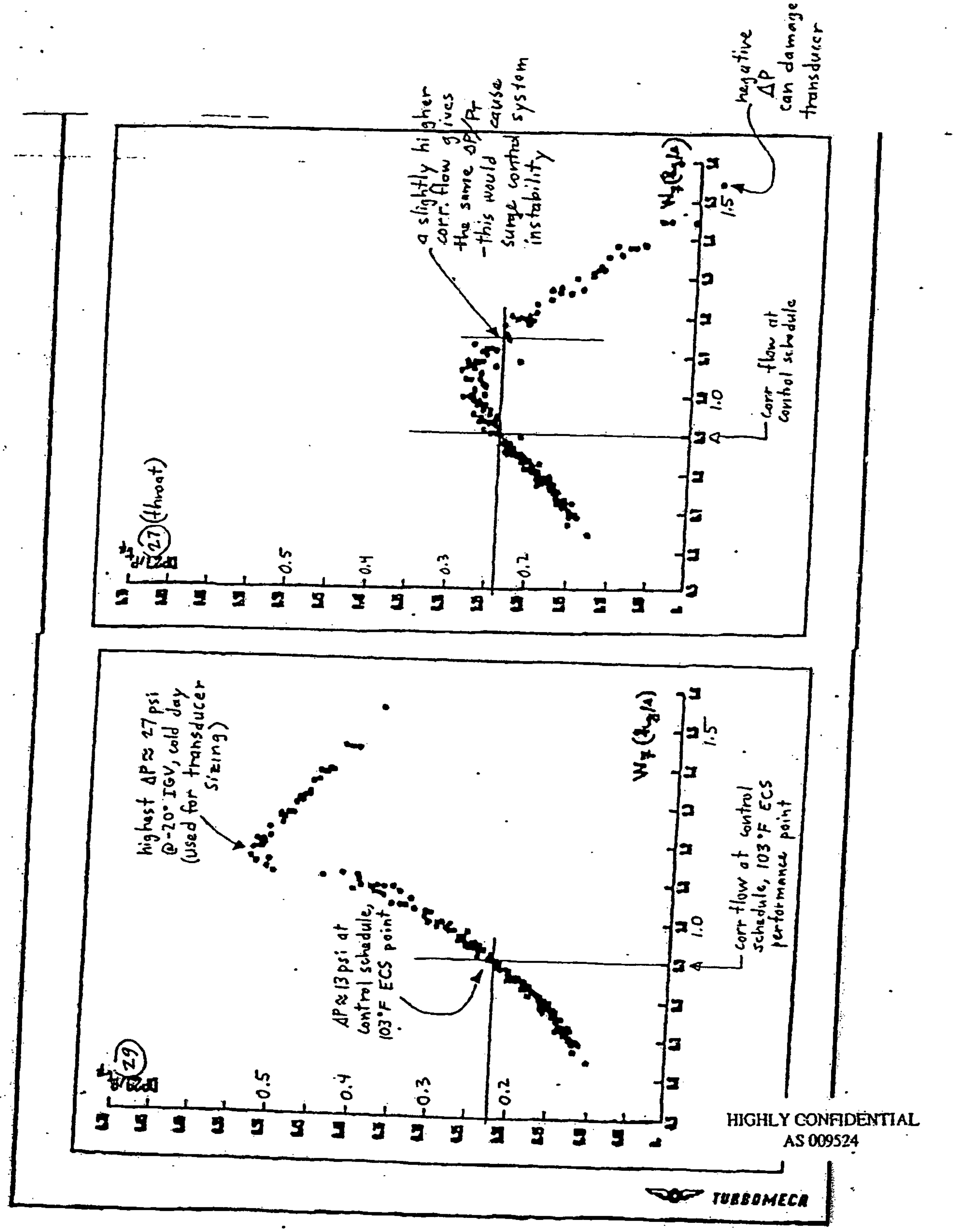
GTCP331-350 APU
COORDination MEMO

CONTINUATION PAGE 2 OF 2

Could Turbomeca please deliver all future load compressors (beginning with S/N 4) with the static port located at position Number 29? We understand that for L/C S/Ns 1, 2, and 3 already delivered to GAPD, the static pressure location is at position Number 27. We would like to rework these diffusers here at GAPD in Phoenix to position Number 29. This will allow us to continue our test and evaluation of the surge control system. Could you please provide the hole position dimensioning for position Number 29 as soon as possible so that we could begin this rework. L/C S/N1 will be disassembled for inspection during the week of May 9 and this would be an ideal time to rework it.

Thanks for your help in this area.

AlliedSignal


DP27/P
27 (throat)
0.5
0.4
0.3
0.2
1.0
1.5
a slightly higher corr. flow gives the same ΔP/PT
–this would cause surge control system instability
negative ΔP can damage transducer
corr flow at control schedule
DP29/P
29
0.5
0.4
0.3
0.2
1.0
1.5
highest ΔP ≈ 27 psi @ -20° IGV, cold day (used for transducer sizing)
ΔP ≈ 13 psi at control schedule, 103°F ECS point
corr flow at control schedule, 103°F ECS performance point
TURBOMECA

DTX 217

The Effect of Rotor Blade Wakes on Centrifugal Compressor Diffuser Performance—A Comparative Experiment

SAMY BAGHDADI

Senior Research Engineer,
Detroit Diesel Allison Division,
General Motors Corporation,
Indianapolis, Ind.

A vortex nozzle facility for testing radial vaned diffusers independently of any rotor has been developed [1,2].[1] This paper describes a comparative experiment designed to evaluate the applicability of results obtained on this facility to actual rotating compressors. Geometrically scaled diffusers were tested in the vortex nozzle facility and in an actual rotating compressor rig, and the results are compared and shown to be very similar in terms of both performance and stability limits. The implications of these results are that blade wake mixing and unsteadiness do not significantly affect diffuser performance.

Introduction

The flow at the outlet of a radial flow impeller is highly swirling, transonic and unsteady in the absolute frame. The unsteadiness is due to the presence of rotor blade wakes. These wakes mix out because of the difference in flow angles between the wakes and the free stream as shown in Fig. 1; however, the rapidity and extent of the wake mixing are presently controversial ([1] discussions). Moreover, the influence of unmixed blade wakes on the performance of the radial diffuser is not understood. This paper attempts to shed some light on the effects of rotor blade wakes on the performance of radial diffusers by presenting data from a comparative experiment.

A unique vortex nozzle facility has been conceived and developed to simulate the essential features of the flow at the outlet of a high pressure ratio centrifugal compressor impeller [2]. This facility produces wake-free, transonic, swirling flow at the inlet to the radial diffuser. The axial (hub-to-shroud) flow profile has been shown to be similar to those measured in compressor practice (authors' closure, [1]). Thus the significant difference between the flow produced by the vortex nozzle facility and that existing at the discharge of actual compressor impellers is due to the presence of wakes in the latter case. The effects of these wakes can be examined by comparing results obtained on the vortex nozzle facility to those obtained on an actual compressor.

Vortex Nozzle Facility

The operation of the vortex nozzle is based on conservation of angular momentum. A swirling flow is accelerated to supersonic speed by decreasing the radius. If the radial component of velocity is small compared to the tangential component, the radial Mach number will be low. Then it is possible at the smaller radius to turn the flow radially outward, maintaining its supersonic tangential Mach number and low flow angle. Fig. 2 is a scale cross section drawing of the vortex nozzle and diffuser test rig. Air flow enters through 30 inlet swirl vanes, and swirls through the vortex tube in a spiral of decreasing radius before it enters the diffuser test section. Both Mach number and flow angle at the diffuser inlet may be varied by changing the inlet swirl vane (ISV) setting angle and mass flow rate. For a given inlet swirl vane angle, flow conditions in the diffuser are regulated by throttle and by-pass valves which control the pressure in the diffuser exhaust plenum (air flow through the rig was induced by a steam ejector located at the extreme discharge end of the flow system). Fig. 3 is a photograph of an early[2] configuration of the rig.

Vortex Rig

Instrumentation. The air flow rate through the vortex nozzle diffuser was measured by means of a standard ASME sharp-edged orifice plate located in a flowmeter duct downstream of

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[1] Numbers in brackets designate References at end of paper.

Contributed by the Fluids Engineering Division of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS and presented at the Gas Turbine and Fluids Engineering Conference, New Orleans, La., March 21–25, 1976. Manuscript received at ASME Headquarters, January 5, 1976.

[2] The tube labeled "Removable Spacer" in Fig. 3 was found to produce undesirably radial and uniform flow at the test section inlet and was therefore removed during the rig shake-down tests [3].

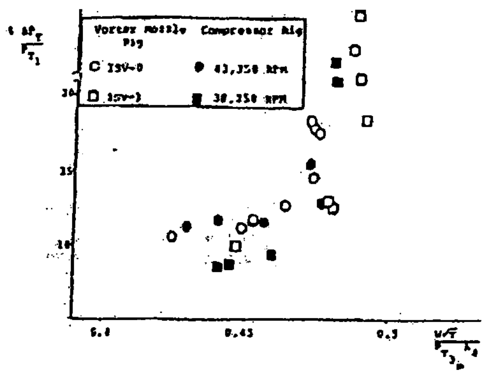

Fig. 15 Diffuser total pressure loss

- The accuracy of the flow measurements on the two rigs is approximately ± 0.5 percent, so that a 1 percent difference in flow between the two rigs is within measurement accuracy.
- The transducer accuracies (typically ±0.05 percent of full scale) can translate into an error in C_p of the order of ±2.5 percent.

The results presented in Figs. 10–15 for the two rigs can thus be said to agree within the range of experimental accuracy.

Conclusions

Since we believe that the only potentially important aerodynamic difference between the two rigs is due to the presence of rotor blade wakes in the compressor rig, the close agreement of the diffuser performances implies that these wakes do not significantly affect diffuser performance, at least for the case of this typical aircraft compressor.

The reason that the rotor balde wakes do not affect diffuser performance can be due to:

- Very rapid mixing of the wakes
- Lack of response of the diffuser vanes to the high frequency flow variations imposed by the rotating wakes
- A combination of the two items above.

The phenomenon of airfoil insensitivity to high frequency flow changes has been encountered in the case of axial compressors operating with circumferential inlet flow distortion, and in the case of helicopter rotor blades undergoing rapid angle of attack changes. In these cases, the airfoil response is found to be an inverse function of the reduced frequency $\bar{w}c/2U$, where $\bar{w}$ is the disturbance frequency, c is the blade chord, and U is the rotational velocity of the airfoil. In addition, Smith and Kline [4] have demonstrated that axial flow diffusers are only sensitive to disturbances having frequencies much lower than the blade passing frequencies of most aircraft engine compressors.

Recommendations

Some of the concern regarding the effect of rotor-blade-wake-induced flow oscillations on diffuser performance and stability is due to the recently discovered flow unsteadiness which precedes surge in centrifugal compressor diffusers. However, work with the vortex nozzle diffuser rig has demonstrated that at least some of this behavior is rotor-independent; unsteadiness of the throat static pressures has been recorded on the vortex nozzle diffuser as the diffuser is loaded prior to surge (these data are presented in the authors' closure to [1]). Moreover, measured presurge diffuser unsteadiness occurs at frequencies much lower than blade passing frequencies [5].

It is recommended that further and extensive dynamic pressure investigations of the surge flow phenomena be conducted using the vortex nozzle diffuser rig. These investigations should reveal whether rotating stall in the vaneless space precedes surge, and help identify the causes of flow breakdown and unsteadiness.

References

1 Baghdadi, S., and McDonald, A. T., "Performance of Three Vaned Radial Diffusers with Swirling Transonic Flow," JOURNAL OF FLUIDS ENGINEERING, TRANS. ASME, June 1975, pp. 155–160.

2 Baghdadi, S., "Study of Vaned Radial Diffusers Using Swirling Transonic Flow Produced by a Vortex Nozzle," PhD thesis, School of Mechanical Engineering, Purdue University, Lafayette, Ind., Dec. 1973.

3 Korn, J. A., "Compressor Distortion Estimates Using Parallel Compressor Theory and Stall Delay," *AIAA Journal of Aircraft*, Sept. 1974, pp. 584–586.

4 Smith, C. R., Jr., and Kline, S. J., "An Experimental Investigation of the Transitory Stall Regime in Two Dimensional Diffusers Including the Effects of Periodically Disturbed Inlet Conditions," Report PD-15, Thermosciences Division, Mechanical Engineering Dept., Stanford University, Stanford, Calif., 1971.

5 Toyama, K., "An Experimental Study of Centrifugal Compressor Surge," M.Sc. thesis, Iowa State University, published as Creare Technical Note TN-168, Hanover, N. H., 1973.

DISCUSSION

W. Fister[1] and H.-P. Muller[1]

Dr. Baghdadi's comparative experimental investigations are of great importance for research concerning the diffusers of centrifugal compressors, as they show that results obtained at simple stationary models may by all means be applied to the conditions in a compressor stage.

It is of particular interest that the pressure recovery coefficient C_{p2-4} is only influenced to such a small extent that a good correlation can be achieved here as well, even with regard to the critical section of impulse exchange behind the impeller tip. As far as the diffuser itself is concerned it is to be expected, however, that the diameter ratio D_3/D_2, i.e. the distance between the diffuser inlet and the impeller outlet, is of importance.

During the past few years we carried out experiments at conical diffusers of an eight nozzle scroll, in order to find out if the efficiency and the pressure recovery coefficients obtained with undisturbed, symmetrical fluid flow in diffusers can also be obtained behind an impeller and a scroll bend. Although our investigations were carried out in the field of incompressible fluid flow and had an aim slightly different from the experiments in question, part of the results point into the same direction and thus should be indicated here:

Fig. 16 shows the principle of the compressor stage with the

[1]Ruhr-Universität Bochum, West Germany.

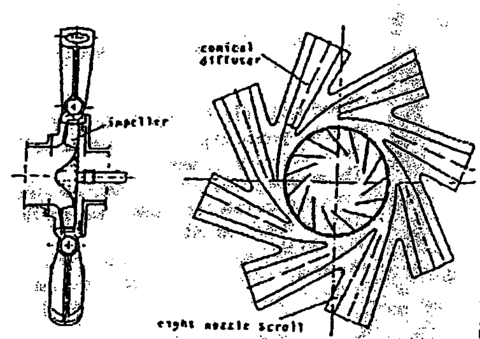

Fig. 16 Compressor rig with eight nozzle scroll

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47 St., New York, N.Y. 10017

82-GT-10

The Performance of Centrifugal Compressor Channel Diffusers

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Test results pertaining to the characteristics of single-stage centrifugal compressors with backswept impellers and channel-type diffusers are presented and analyzed to formulate major performance criteria influencing maximum diffusion capability. For any given stage, it was determined that stage surge (when triggered by diffuser stall), occurred near a constant mean stream velocity diffusion ratio between the impeller tip and diffuser throat. This diffusion ratio attained a maximum value of 1.8 for impeller tip Mach numbers less than unity, but was not unique for all stages, being more intimately coupled with throat blockage accumulation as a function of diffusion rate. This was identified by testing some vaned diffusers beyond the stall limit where rapid blockage accumulation precipitated an immediate decrease in channel diffuser and system static pressure recovery. The results of various experiments in the vaneless space are also described to illustrate the sensitivity of the vaneless space flow upon centrifugal compressor performance.

NOMENCLATURE

A	Area
AS	Aspect Ratio (=b/w)
b	Vane Height
B	Blockage Factor
B_n	Normalized Blockage Factor
C_p	Static Pressure Recovery
C_D	Discharge Coefficient
C	Absolute Velocity
D	Diameter
i_3	Throat Incidence
L	Vane Length
N	Rotational Speed
P	Total Pressure
p	Static Pressure
q	Impeller Work Factor
T	Total Temperature
U	Impeller Tip Speed
W	Flow, Throat Width, Relative Velocity
Z	Vane Number
δ	Boundary Layer Displacement Thickness
α	Absolute Flow Angle
η	Efficiency

SUBSCRIPT

1	Impeller Inlet
2	Impeller Tip
3	Diffuser Throat
4	Diffuser Exit (covered)
E	Exit
R	Radial

SUPERSCRIPT

*	Unblocked Velocity

UNITS

The following units were used in evaluation of the compressor performance. Equivalent metric conversions are noted.

Item	English	Metric Conversion
Ns	rpm $(cfs)^{0.5}$ X $Had^{-0.75}$	1 rpm $(M^3/S)^{0.5} m^{-0.75}$ = 0.412 X Ns
C; U; W	fps	1 m/s = 3.281 fps
D; L; r; t	in.	1 cm = 0.3937 in.
$\alpha 1$; $\alpha 2$	deg	---
ρ	lb/ft^3	$1 kg/m^3$ = 0.0625 lb/ft^3
cp	Btu/lb	1 J/kg = 0.43 X 10^{-3} Btu/lb
P;p	psia	1 kg/m^2 = 1.422 X 10^{-3} psia
T	deg R	deg K = deg R/1.8

Contributed by the Gas Turbine Division of the ASME.

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INTRODUCTION

Flow ranges for single-stage centrifugal compressors are dictated by the stalling characteristics of the impeller and the diffuser which are intrinsically controlled by the diffusion capability or attainable static pressure rise of the blade and vane rows. Although both vaned and vaneless diffuser systems are used for centrifugal compressors, the requirement for maximum efficiency at high Mach numbers makes the use of vaned diffuser systems almost mandatory. The impeller and diffuser must be matched simultaneously at their peak efficiency flow conditions.

The stationary vaned diffuser tends to be the flow controlling component in that its overall Mach number level and inlet blockage are higher than those of the inducer which operates with a large radial variation of Mach numbers from hub to shroud. The diffuser must also accept an already diffused flow from the impeller with resulting non-uniform entrance conditions which further aggravate its stalling sensitivity. These conditions curtail the compressor operating range and, as a result, stationary diffusers for centrifugal compressors have received considerable attention. Attainment of a large flow range requires that the impeller and the diffuser must be capable of extended operation into their stalled or positive incidence regions to a flow where static pressure rise plateaus, and compressor surge is eventually triggered. Stage surge is believed to stem from operation on an unstable (positive slope) portion of the overall compressor characteristic, where the static pressure ratio increases with increasing flow. One effective method of increasing compressor operating range is to provide sufficient impeller stability so that the downstream diffuser can operate slightly into its positive incidence zone, even though the diffuser static pressure recovery versus flow characteristic exhibits a positive slope.

Previous studies by the author on centrifugal impeller diffusion limitations were presented in Reference [1] and pertained to analysis of a single, experimental, high Mach number, centrifugal, backswept impeller of near optimum configuration. Test results on this particular impeller indicated impeller stalling occurred whenever the relative velocity diffusion ratio, W_{IRMS}/W_2, (based on mixed impeller exit condition) exceeded 1.6. It was, therefore, suggested that such a simple limiting velocity ratio could be used as an initial design guideline to indicate impeller stalling proximity. The informative discussion in Reference [1] added a precautionary tone in that application of such a simple stalling proposition to all centrifugal impeller designs might be premature. Bearing this precaution in mind, additional research was conducted on the stalling characteristics of a wide variety of centrifugal impeller designs, mostly of the inducer type with back-sweep angles of 40-45 degrees, relative to the radial direction).

The goals in conducting this additional research were continued improvement in performance levels, performance prediction techniques, and, particularly, the identification of key factors improving stable operating flow range of both the impeller and diffuser. Analysis of the impeller performances suggested that a modified diffusion factor, including the effects of meridional curvature and blade solidity, provided improved

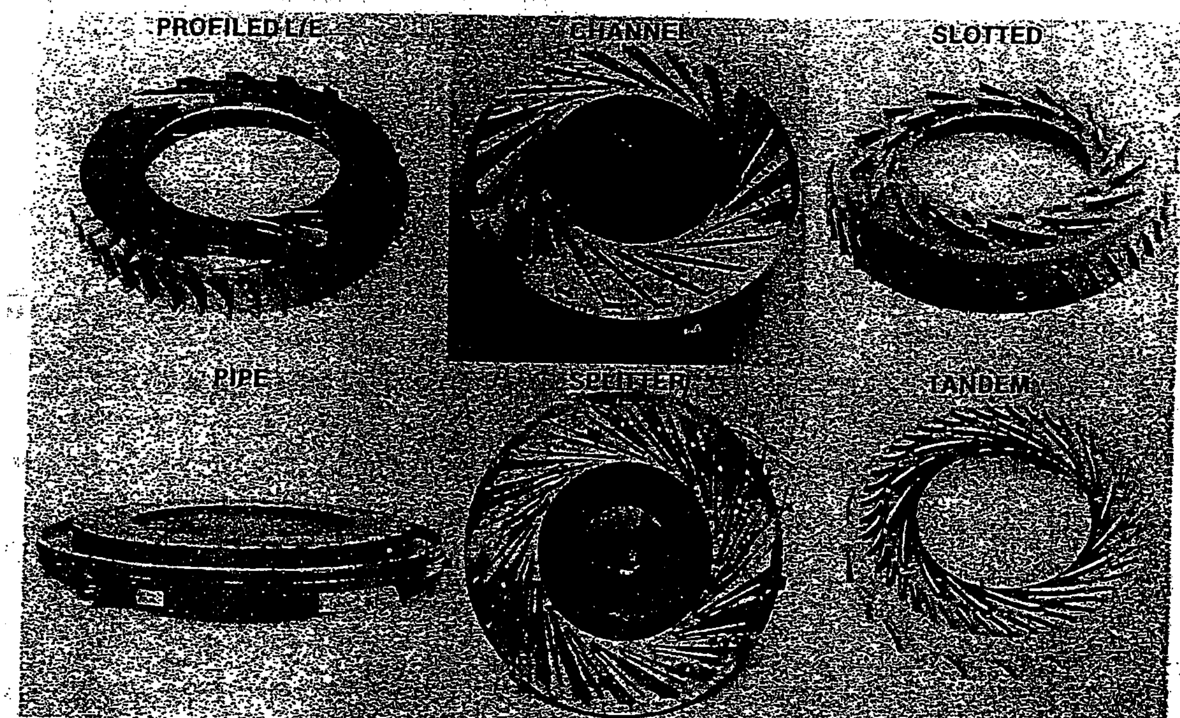

Fig. 1 Types of centrifugal compressor diffusers

pressure, is more fundamentally determined by the vaneless space diffusion ratio C_2/C_3. Thus, even in length limiting diffuser installations, it is possible to obtain a respectable overall diffuser static pressure recovery level.

The justification for using a mean stream diffusion ratio approach in such a complex flow pass, dominated by endwall and flow mixing effects, comes primarily from convenience. The intent was to derive a simplified stall parameter that could be used in the preliminary design phase prior to detailed internal flow analysis. More sophisticated fluid dynamic models capable of predicting boundary layer growth, corresponding viscous shear and mixing losses, and separation onset in highly unsteady-flow may eventually be derived for vaneless space flow characterization. For the immediate future, more experimentation as typified by the extensive work described herein is prescribed: hopefully, with improved instrumentation techniques.

The experimental work necessary to provide an improved understanding should encompass investigating several flow stages from low to high specific speed operating both in and out of the surge or stalling zones.

ACKNOWLEDGEMENT

The author wishes to acknowledge the efforts of R. Geiser, Senior Development Engineer who assisted in the computer analysis of the test data.

REFERENCES

1 Rodgers, C., "Impeller Stalling as Influenced by Diffusion Limitations," ASME Journal of Fluid Mechanics, March 1977

2 Rodgers, C., "A Diffusion Factor Correlation for Centrifugal Impeller Stalling," ASME 78-GT-61

3 Blair, L.W., and Russo, C.J., "Compact Diffuser for Centrifugal Compressors," A.I.A.A. -80-1077

4 Rundstadler, P.W., Jr. and Dean, R.C., Jr., "Straight Channel Diffuser Performance at High Inlet Mach Numbers," ASME Journal on Basic Engineering, Vol. 91, No. 3 September 1969

5 Came, P.M., Herbert, M.V., "Design and Experimental Performance of Some High Pressure Ratio Centrifugal Compressors," AGARD -CP-282, 1980

6 Yoshinaga, Y., Gyobu, I., Mishina, H., Koseki, F., Nishida, H., "Aerodynamics Performance of a Centrifugal Compressor with Vaned Diffusers," ASME Journal of Fluids Engineering, December 1980

7 Krain, H., "A Study of Centrifugal Impeller and Diffuser Flow," ASME 81-GT-9

8 Kenny, D.P., "A Novel Correlation of Centrifugal Compressor Performance of Off-Design Prediction," A.I.A.A. -79-1159

9 Rodgers, C. and Sapiro, L., "Design Considerations for High Pressure Ratio Centrifugal Compressors," ASME 72-GT-91

10 Greitzer, E.M., "Review of Axial Compressor Stall Phenomena," ASME Journal of Fluids Engineering, June 1980

11 Conrad, O., Raif, K., Wessels, M., "The Calculation of Performance Maps for Centrifugal Compressors with Vane-Island Diffusers," Performance Prediction of Centrifugal Pumps and Compressors, ASME 1980

12 Reneau, L.R., Johnston, J.P., and Kline, S.J., "Performance and Design of Two Dimensional Diffusers," ASME Journal of Basic Engineering, Vol. 89, March 1967

13 Japikse, D., "The Influence of Diffuser Inlet Pressure Fields on the Range and Durability of Centrifugal Compressor Stages," AGARD -CP-282, 1980

14 Rodgers, C., "Variable Geometry Gas Turbine Radial Compressors," ASME 68-GT-63

DTX 287

TN - 186

TECHNICAL NOTE

DIFFUSER DATA BOOK

Peter W. Runstadler, Jr.

Francis X. Dolan

Robert C. Dean, Jr.

TN - 186
MAY 1975

Creare INC.
SCIENCE AND TECHNOLOGY

HANOVER, NEW HAMPSHIRE

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2 OBJECTIVES

The purpose of this document is to provide the fluids engineer ready access to this extensive collection of diffuser characteristics. A second objective is to aid the designer in correctly applying this information.

However, this document has not been prepared to address the detailed fluid-physics discussion of the nuances of diffuser behavior. For an exhaustive treatment in that regard, see the source documents behind this summary. For a succinct overview, refer to:

> McDonald, A. T. and Runstadler, P. W., Jr.; BEHAVIOR OF DIFFUSERS WITH DISTURBED AND UNSTEADY INLET CONDITIONS; Presented at Symposium on Fluid Dynamics of Unsteady Three-Dimensional and Separated Flows, Project SQUID, Office of Naval Research, The School of Aerospace Engineering, Georgia Institute of Technology, Atlanta, June 10, 11, 1971.

3 TECHNICAL HISTORY

The first known diffusers were designed to cheat the authorities.* In Rome (circa 100 A.D.) water was distributed to homes of the wealthy via an aqueduct system. The charge for the water was based upon flow rate, because the water ran continuously. At each outlet, the flow water was metered by an ajutage which amounted to a long flow-metering nozzle made of lead. Some enterprising Roman discovered that if he flared the adjutage, his flow rate increased without increasing his cost! Hence, from necessity and avarice, the diffuser was invented.

The rational appreciation and application of diffusers probably did not commence until the 18th century, when hydraulic engineers began to understand that a significant portion of pumping power can appear in a system as flow kinetic energy. Leonhard Euler (1707-1783) showed the mathematical relationship between flow kinetic energy, elevation and pressure.** From that theoretical cornerstone, diffuser technology has been constructed for over 200 years.

The history of diffuser technology seems remarkable, but is perhaps a typical example of how man's knowledge structures are really built. Until recent times, there were two divergent schools at work. The theoreticians have attempted to predict diffuser performance analytically, for two centuries and are still trying, but without much success.

The practical engineer built and tested diffusers for his particular needs. A few pragmatically oriented researchers tested families of diffusers. None of these empiricists made a systematic search, exploiting the theoretical tools as they could, in order to identify the key parameters and the characteristic behavior modes of the diffuser.

Those who have studied the history of hydraulics know that there was a classical period in the late 18th and in the 19th century when leading thinkers thought that theory alone was going to explain everything. At the turn of the 20th century, there were famous, now slightly amusing, arguments between the theoreticians and the pragmatists over such subjects as whether the bumblebee could fly. The existing theory could not explain the lift on a wing, but everyone with common sense knew that wings could lift

*Rouse and Ince (1957), p. 28.
**Rouse and Ince (1957), p. 104.

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Further Data on the Pressure Recovery Performance of Straight-Channel, Plane-Divergence Diffusers at High Subsonic Mach Numbers

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Measurements are reported of the pressure recovery of straight-channel, symmetric, single-plane-divergence diffusers with inlet Mach numbers between 0.2 and choking for an aspect ratio of 5.0. The data reported cover a range of length-to-throat width ratios L/W_1 and divergence angles 2θ for diffuser geometries near peak recovery. These data complement data previously reported for $AS = 0.25$ and 1.0. Diffuser performance maps are given that show pressure recovery C_p as a function of diffuser geometry for fixed values of throat Mach number M_1, throat blockage B, and aspect ratio AS for the range of variables tested. Of significant importance to the designer is the alteration in the shape of the pressure recovery contours on the performance maps with variations in M_1, B, and AS.

Also reported are data on the effect of changes in diffuser inlet Reynolds number, asymmetric distribution of inlet blockage around the throat periphery, and the influence of rounded throat corners on the pressure recovery behavior of the straight-channel diffuser. These data have underscored the necessity of understanding the cumulative effects of a number of secondary parameters on pressure recovery.

The importance to the designer of a knowledge of how diffuser performance depends upon the diffuser geometric and inlet parameters is discussed.

Introduction

The diffusing passage is a key element of many fluid machines and fluid dynamic systems. The ability to recover pressure and/or the ability to establish a stable flow or a flow of low distortion is critical to the behavior of many devices and systems which incorporate a fluid dynamic diffuser. The optimum performance and the proper design of diffusing passages for many devices, for example in turbomachines, aircraft inlets, carburetors, flowmeters, noise suppressors, etc., depends upon an understanding of the important flow parameters governing the performance of the fluid dynamic diffuser.

In an earlier paper (reference [1]),[1] pressure recovery performance data for straight-channel, single-plane-divergence diffusers at high subsonic inlet Mach numbers were presented. The

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[1] Numbers in brackets designate References at end of paper.

Nomenclature

A = boundary layer suction surface area

$A_{effective}$ = equivalent one-dimensional flow area

$A_{geometrical}$ = actual diffuser inlet area

AR = area ratio = $1 + 2(L/W_1) \tan\theta$

AS = aspect ratio = b/W_1

b = diffuser depth

B = blockage factor $= 1 - \dfrac{A_{effective}}{A_{geometrical}}$

C_p = static pressure recovery coefficient $= \dfrac{P_e - P_1}{P_{01} - P_1}$

C_Q = boundary layer suction t[o] efficient $= \dfrac{v_s}{U_\infty}$

D = hydraulic diameter $= \dfrac{2bW}{b + W}$

k = ratio of specific heats

(Continued on next page)

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DTX 293

The Dynamics and Thermodynamics of COMPRESSIBLE FLUID FLOW

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IN TWO VOLUMES

VOLUME I

THE RONALD PRESS COMPANY · NEW YORK

DTX 295

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The Use of Compressor-Inlet Prewhirl for the Control of Small Gas Turbines

The objective of this paper is to describe a control system recently developed to provide nearly instantaneous power response with a two-shaft gas turbine. The principal elements in this system are movable guide vanes in the compressor inlet. The effects of air prewhirl on centrifugal compressor and gas-turbine engine performance are discussed and test results are presented.

Introduction

Nearly instantaneous response to rapid loading conditions has been developed for the two-shaft turbine. This response is in the order of ten millisec and has been accomplished without sacrificing any of the performance advantages of the two-shaft machine.

An analysis revealed that by providing prewhirl to the compressor inlet air the horsepower of a two-shaft engine could be varied without changing gas producer speed.

Prewhirl is produced by giving the incoming air a whirl component through the use of inlet guide vanes. A positive vane angle produces prewhirl in the direction of impeller rotations and a negative vane angle produces prewhirl in the opposite direction. Air prewhirl in the direction of impeller rotation reduces compressor work at a given rpm. Air prewhirl counter to impeller rotation increases compressor work.

In this manner engine pressure and temperature level can be varied to change output power without having to change compressor rpm.

Discussion

Compressor Work Variation With Prewhirl. Fig. 1 represents the mean velocity triangle at the impeller inlet while Fig. 2 represents the equivalent velocity triangle at the impeller exit. From the Euler equation, assuming simple one-dimensional flow theory, the theoretical amount of work imparted to each pound of air as it passes through the impeller is given by:

$$W = \frac{U_2 V_{u2} - U_1 V_{u1}}{g_c} \tag{1}$$

where

W = Work per lb of air
U_2 = Impeller peripheral velocity
U_1 = Inducer velocity at the mean radial station
V_{u2} = Absolute tangential air velocity at impeller exit
V_{u1} = Absolute tangential air velocity at inducer inlet
g_c = Gravitational constant

Both the work input and the compressor isentropic efficiency determine the pressure ratio produced by a compressor with given inlet condition.

In a typical centrifugal compressor, without inlet guide vanes,[2] air enters the compressor axially, i.e., V_{u1} is zero. The compressor work capacity then reduces to

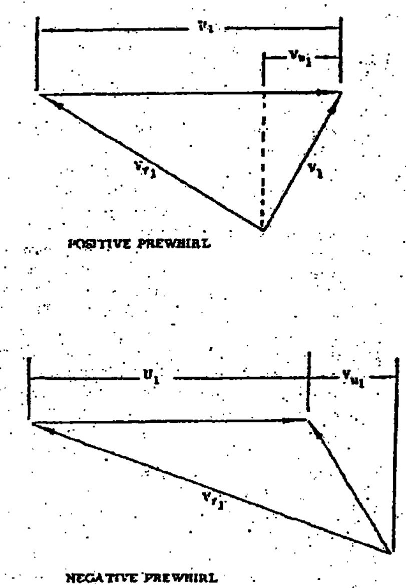

Fig. 1 Inlet velocity triangles

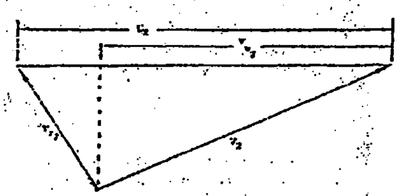

Fig. 2 Exit velocity triangles

$$W = \frac{U_2 V_{u2}}{g_c} \tag{2}$$

It is seen from this expression that W is only a function of U_2 and V_{u2} or, in the ideal case with purely radial vanes, the air will leave the impeller tip with a tangential velocity equal to the impeller peripheral velocity and the work becomes only a function of compressor rotational speed. Accordingly, for a constant

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DTX 305

TYPICAL COMPRESSOR CONTROL CONFIGURATIONS

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ABSTRACT

A centrifugal compressor control system generally consists of two parts: a loop to control a process variable, such as discharge pressure or gas flow, and a surge control loop to maintain the compressor in its stable operating range. The configuration of the control system is determined by the process variable, the kind of compressor, the size and nature of load changes and the accuracy of surge control required. Examples of control systems designed to meet different requirements are given. Related problems including reset windup in surge controllers, interaction of control loops, manual/automatic switching and override controls are discussed.

INTRODUCTION

The primary concern in a compressor control system is the process variable - the variable in the compressor performance that is controlled to meet process objectives. The process variable control loop is usually straightforward, showing variations primarily in the variable selected and in the final control element. Commonly controlled process variables include:

1. Discharge pressure P_d
2. Suction pressure P_1
3. Mass flow Q_m
4. Related variables - such as temperature in a refrigeration compressor.

The final control element in the process variable loop is determined by the kind of compressor operation. The different kinds of compressor operation to be considered here are:

1. Constant speed with inlet throttling.
2. Variable speed.
3. Constant speed with inlet guide vanes.

The final control element on a constant speed compressor with inlet throttling is usually an air operated control valve in the suction line. For the variable speed compressor, the gas or steam turbine drive can be considered to be the final control element, responding to control signals through a governor. Inlet guide vanes are an integral throttling device in the compressor and are furnished with actuators to respond to conventional pneumatic and electronic control signals.

Surge control is of secondary interest in process operation but is vital for protection of the compressor. Because a centrifugal compressor cannot operate safely below a certain surge flow, any system in which the flow demand can fall below the compressor surge flow must include surge control.

Surge control is simply effected by opening a bypass valve around the compressor or blowing off gas to atmosphere to maintain minimum flow through the compressor. Since bypassing or blowing off gas wastes power, it is desirable to determine surge flow as accurately as possible to avoid bypassing gas unnecessarily while maintaining safe operation. However, determining surge flow is often not a simple matter. Surge flow for a compressor is not a fixed quantity, but is related to other variables. Where other variables substantially affect surge flow, they must be measured and included in the surge system. In fact, surge conditions can be defined completely in terms of variables other than flow. The problem of defining surge conditions has led to the development of a wide variety of systems for determining surge conditions from different measurements.

The surge control system selected for a particular application will depend on a number of factors, including:

1. Kind of compressor - a constant speed compressor has one less variable affecting surge conditions than a variable speed machine, and determination of surge conditions is usually simpler.
2. Load changes - where load conditions, such as inlet temperature or discharge pressure are relatively constant, these variables can be treated as constants for surge control, resulting in a simpler surge system.
3. Ease and availability of measurements - where a measurement is already being made for other purposes, a surge system utilizing this measurement will be more economical. Where one kind of measurement is impractical (flow measurement, for example, may produce objectionable permanent pressure loss) a surge system utilizing another kind of

permit manual start and emergency operation in the field, while observing and listening to the compressor. The problem with a system of this kind is transferring control back and forth from the field to the control room without upsetting control. Fortunately, several modern electronic and pneumatic control stations include features to make the transfer possible without balancing controllers.

In the system shown, the field M/A station tracks the signal from the control room continuously and can be switched to manual at any time, without balancing by the operator. When the field station is on manual, a logic standby signal is sent to the control room, causing the control room stations, whether on automatic or manual, to track the valve position. The field station can be switched back to automatic, transferring control back to the control room for normal operation, also without balancing.

Figure 23 - In some situations, it is desirable not to allow the operator full manual control of the surge valve, so that he cannot inadvertently put the compressor into surge. Figure 23 shows a circuit in which the operator can only open the blow off valve, and not close it beyond the position dictated by a surge controller. The manual control for the surge loop consists of a manual loading station and a low selector relay, which will permit the operator to open the blow off valve. The external reset feedback signal to the surge controller is included to prevent reset windup when the valve is under manual control. The pressure controller includes the standard manual/automatic switch in the control station.

CONCLUSION

A considerable part of this paper has been devoted to different kinds of surge control systems, derived from a variety of measurements, but all based on the theoretical properties of the surge line. It will be found in many cases that the surge lines on the compressor maps furnished by compressor manufacturers do not conform exactly to the theoretical properties. One reason for this is that surge usually occurs in one stage of a multi-stage compressor, and the inlet and discharge conditions may not reflect the conditions at the stage in which surge occurs. Nevertheless, a surge control system based on the theoretical surge line will be correct as to direction and can be calibrated to give the required protection. Unless the performance of the compressor is well established from an identical design, a surge control system should include simple gain and bias adjustments to permit adjusting the position of the surge line as dictated by field experience.

REFERENCES

(1) Boyd, David M., "How Accurate are Surge Curves?" Instrumentation in the Chemical and Process Industries, Vol. 10, ISA, pp 35

(2) Gaston, J. R., "Centrifugal Compressor Control", Instrumentation in the Chemical and Process Industries, Vol. 10, ISA, pp 1

(3) White, M. H., "Surge Control for Centrifugal Compressors", Chemical Engineering, 5/25/72